

TURBOMACHINERY BALANCING CONSIDERATIONS

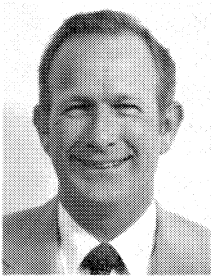
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ABSTRACT

The mechanics of balancing, including details concerning tolerances and specifications, tooling, balancing machines, and procedures have been presented many times in the past. However, very little information has been offered to the user on how to apply these details, or what to consider when one encounters those "other than textbook" circumstances that commonly occur with turbomachinery rotors.

INTRODUCTION

The main goal of a turbomachine turnaround is to start up on schedule, without any problems related to the machine, and hopefully, with an operating speed vibration of approximately 0.5 mils, or less. To achieve this goal on a consistent basis requires dedicated and technically capable people in the reliability group, field maintenance, and repair center organizations. It also requires a concerted effort between these organizations, proper execution of good guidelines and procedures, and sound technical judgement in many diverse areas.

From this presentation, the reader will be assisted in making sound decisions as applied to balancing of turbomachinery rotors. Common causes of unsatisfactory operating speed vibration due to unbalance are discussed, along with practical ways to identify and solve potential problems during rotor repairs.

VIBRATION PROBLEMS RELATED TO ASSEMBLY AND BALANCING

Most problems that occur during assembly and/or balancing of turbomachinery rotors can be attributed to inexperience, over-

sights, or lack of sound judgement. Some of the most common mistakes are made in one or more of the following categories:

- Inadequate inspection of rotors
- Balancing a centerline "set" in an assembled rotor
- Excessive top clearances on component keys
- Excessive stresses induced by component design and shrink fits
- Insufficient shrink fit of components
- Correction of unbalance in improper planes
- Excessive unbalance left in rotor during balancing
- Balancing on areas other than bearing journals

Inadequate Inspection of Rotors

A thorough discussion of rotor inspection techniques would lead far from the intent herein. However, a few points must be addressed in order to allow an adequate discussion of balancing considerations.

Documentation of Runout

Over the years, different repair centers have each come up with their own "standard" method of documenting runouts, consequently, there are many "standards." Rather than complicate this issue, the proper way to document runouts will be discussed, and an explanation of why this method is proper will follow. The following steps are necessary:

- Place the rotor in vee blocks, on its bearing journals.
- Select a "zero" reference (coupling keyway, thrust keyway, etc.).
- Use an indicator graduated in tenths of thousandths.
- Roll the rotor in the direction of normal rotation.
- Record the maximum indicator reading.
- Record the phase angle of the maximum reading, referenced to the established "zero."

The above steps assure that sufficient information is acquired to enable sound balancing and/or repair judgement. This method of documenting runouts enables the user to determine maximum centerline deflections, as well as the location of the deflections. Phase information gathered while rolling the rotor in its normal direction allows comparisons of online vibration data to shop runout data. It also allows a clear picture of the shaft centerline shape. In short, runout numbers without phase are not very useful.

A common method of documenting runout is the four point method, where the indicator reading is recorded every 90 degrees (or at three, six, and nine o'clock). This method does not allow accurate centerline shape determination, and in some cases, may be very deceiving in the amount of runout recorded. In the worst case, where the actual high spot was exactly between the four point readings, one would only detect 70 percent of the true runout using the four point method.

Analysis of Runout Documentation

After rotor runout has been documented as previously described, take the time to study the data. If the runout phase changes more than 120 degrees longitudinally across the bearing journals, the rotor may be bowed. Further, if phase differs by more than 120 degrees between the OD and the face of a compressor impeller suction eye, the impeller is probably cocked on the shaft. If an impeller is cocked badly enough, the shaft may also be significantly deflected by the excessive stresses. Obviously, balancing a rotor with either of these conditions could be a very costly mistake, especially if the rotor operates at high speed and above its first bending critical. Temporary shaft bows and cocked impellers will tend to straighten themselves during the bending critical. If the rotor was balanced with a bow and/or cocked impellers, the amplitudes may be highly magnified when straightening occurs, leading to catastrophic failure.

An example is shown in Figure 1 of properly documented runouts on a compressor rotor. To the untrained eye, the runouts may seem low, and consequently, insignificant. In fact, the runouts are well within most OEM specifications, and most likely, meet most inspectors' expectations. A closer look reveals that impeller number one is most likely cocked on the shaft, since the phase of the runouts on the suction eye OD and face are 170 degrees apart. Further, the large differences in phase across the bearing journals suggest a bowed shaft.

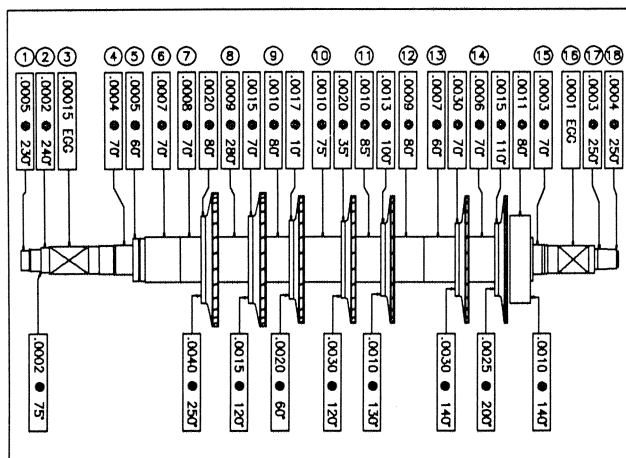


Figure 1. Proper Runout Documentation.

The basic centerline shape of the rotor is shown in Figure 2. This plot was made using the runout information shown in Figure 1, in the following manner:

- Layout the distances between runout points to scale on graph paper. (Numbers 1-18 in Figure 1 correspond to 1-18 in Figure 2).
- Take either the sine (or cosine) of the phase angle, then multiply the result times the amount of runout. (For example, point number 1 is 0.5 mils @ 230 degrees; $\sin 230 \text{ degrees} = -0.766$; $-0.766 \times 0.5 \text{ mils} = -0.38$).
- Plot the result of the preceding step for each runout point of interest.

The final plot will be indicative of the centerline shape of the rotor, however, this method does not provide a plot that is representative of the actual amount of centerline deflection. This kind of information is very helpful during rotor inspections and can help identify areas of excessive shaft stresses as well as bows.

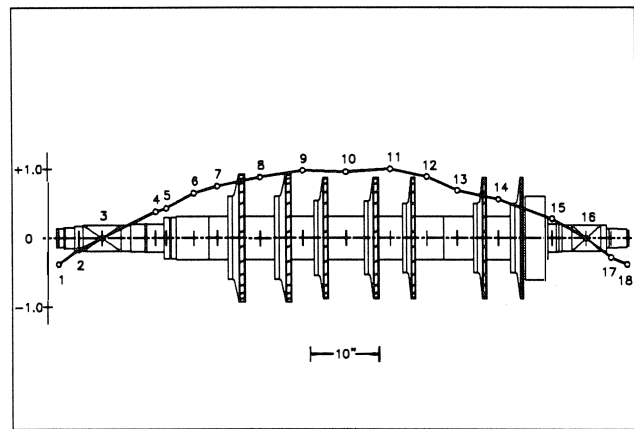


Figure 2. Plot of Rotor Centerline Shape.

Balancing a Centerline Set in an Assembled Rotor

A centerline "set" is a condition in which a rotor's geometric centerline has changed from its normal position either temporarily or permanently. Since temporary sets are those that set the stage for most oversights, these are the ones that will be addressed here. Temporary sets are commonly found in several forms, including:

- Turbine rotors with thermal bows.
- Large, flexible rotors with gravity bows.
- Built up stresses from stacked components.
- Rough handling of stacked rotors.

Turbine Rotors with Thermal Bows

Thermal bows in turbine rotors are quite common. Often, a turbine is tripped and allowed to stop rolling while the rotor is still very hot. Further, on condensing machines, many operators allow the rotor to stop rolling, while still under a vacuum and while sealing steam is still being applied. The sealing steam being admitted to one spot on the shaft will surely result in a shaft bow. The proper way to shut down a turbine is as follows:

- Trip the turbine while running at governor speed to allow the critical range to be rapidly traversed.
- Reset the trip and crack open the throttle valve before the rotor stops rolling.
- Slow roll at 300 to 600 rpm for a minimum of 30 minutes to allow slow, even cooling.
- Break the vacuum and shut off the sealing steam just before closing the throttle valve to stop the turbine.

It must be recognized that while following the shutdown procedure will lessen the degree of a thermal bow, it will not always prevent a bow from occurring. The author sincerely believes that failure to consider and properly address thermal bows in turbine rotors is one of the most common mistakes made during shop balancing practices. A thermal bow may not be detectable with conventional dial indicator readings due to erosion/corrosion pitting of shaft areas, which will not allow accuracy of eccentricity measurement. Further, even if accurate readings are obtainable, the bow may be too small to identify using dial indicators. Consider the following example: 10000 rpm steam turbine rotor—journal weight is 2000 lb. Balance tolerance = $4W/N = 0.8 \text{ oz-in/plane}$. Allowable eccentricity = $0.8 \text{ oz-in}/(2000 \text{ lb} \times 16 \text{ oz}) = 0.000025 \text{ in}$, or 25 $\mu\text{-in}$.

Even if the turbine rotor was four times the balance tolerance (0.0001 in, or 100 $\mu\text{-in}$ of eccentricity), it may be extremely

difficult to detect with dial indicators. However, a good balancing machine can easily detect shaft bows in turbine rotors, and therein lies the problem; many people will simply “correct” the unbalance caused by the thermal bow. These same people will then wonder why the turbine’s vibration levels don’t meet expectation when the rotor is installed and started, and the thermal bow comes out!

The solution to the problem just described is simple. If the rotor was running well prior to shutdown, *do not pull it for balancing*. Secondly, (for those that insist on check balancing a rotor that was operating satisfactorily), always expect a turbine rotor to have a thermal bow. This will normally present itself in the form of a fairly large, static unbalance of four or more times the balance tolerance. In other words, the phase difference between the two end planes will be 15 degrees or less, and the amount of unbalance on each plane will be fairly equal in gram-inch (or ounce-inch) terms.

When a turbine rotor must be balanced, and the preceding symptom of a thermal bow exists, the rotor can usually be straightened by allowing it to continuously slow roll in the balancing machine while it is evenly heated with torches to approximately 350°F to 400°F. Allow the rotor to roll continuously until cool, then run it up again and check the unbalance. If it has significantly improved, the heating procedure should be done once again to assure centerline stability before making the actual corrections. Though this procedure may cost a little more money and an additional eight to ten hours on the balancing machine, it will at least provide assurance that the user does not balance out a thermal bow and be sorry later.

Large, Flexible Rotors with Gravity Bows

Large, heavy rotors with significant bearing centerline spans are subject to gravity bows. Bows of this type will always straighten themselves while the rotor is rolling in the balancing machine. The initial symptoms are identical to the thermal bow symptoms previously discussed; however, no heat will be required for straightening. Large rotors should be continuously slow rolled for 15 minutes between the balance runs until the balancing machine provides repetitive readings. During actual corrections, the rotor’s downtime must be kept to a minimum to prevent further centerline sets. In summary, never attempt balance corrections on a large rotor based on one balance run.

Built up Stresses from Stacked Components

This problem is common on compressor rotors where the design is such that the length of the bore-to-diameter of the bore (L/D) ratio is greater than 0.75, and a fairly heavy and uniform shrink fit is used across the bore. It can be further aggravated by nonuniform heating of the impeller prior to assembly, and/or tapping on the sides of an impeller with a soft hammer in an attempt to “straighten” it before the bore contracts and grabs the shaft.

Consider an impeller with a 13¼ in long, 7½ in diameter bore, that has an interference fit of 0.002 in/in diameter (Figure 3) To install this impeller, it must be heated to provide approximately 0.023 in of bore expansion, or 472°F above ambient shop temperature. At this temperature, the bore length of the impeller will expand by approximately 0.040 in. As the impeller cools and the bore contracts, tremendous stresses are built up in the components by the uniform shrink fits across the bore length. These stresses commonly result in excessive centerline deflections of the shaft. If the stresses are ignored and the components are “balanced,” the result is too often a rotor with unacceptable levels of vibration. As the rotor enters the first bending critical, the stresses will typically relieve somewhat, and the rotor takes on an entirely different shape than it had when it was balanced. When this happens, large unbalances are induced while the rotor is in the critical, resulting in very high vibration that may be catastrophic in nature.

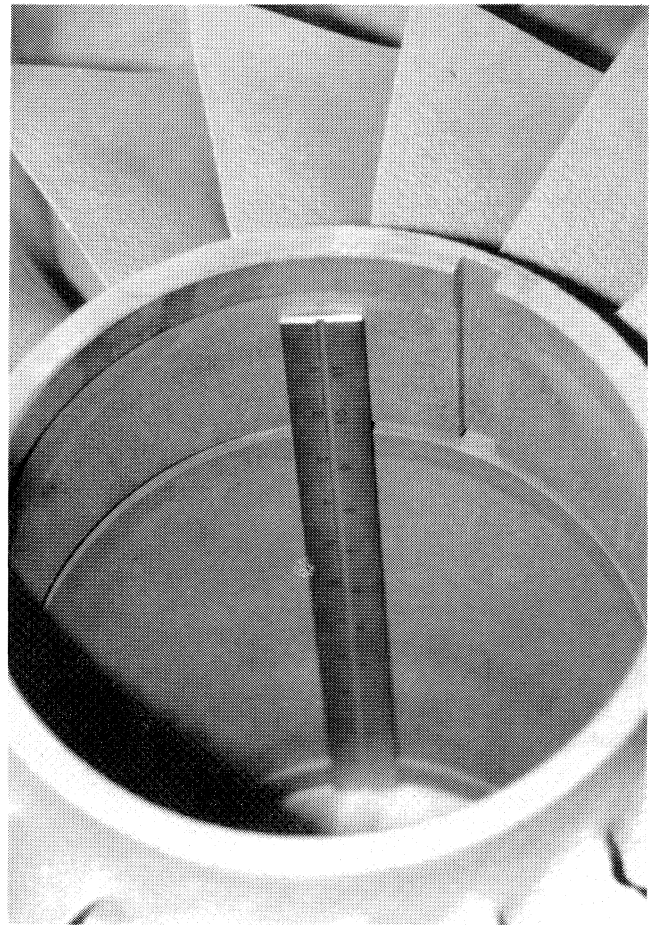


Figure 3. Compressor Impeller with Bore L/D > 0.75.

With some designs, it is impossible to cure this problem, however, much can be done to alleviate it (Figure 4). First of all, it must be realized that most modern compressor impellers require an interference fit of 0.0015 in to 0.002 in/in of diameter on the backplate (discharge) side due to centrifugal forces at speed (depending on impeller diameter and speed). However, the suction side of the typical closed impeller design will normally require an interference of only 0.00075 in - 0.001 in/in diameter at most. When the L/D ratio of the impeller bore is greater than 0.75, the following steps are recommended to alleviate excessive rotor stresses:

- Machine a relieved area in the center of the bore, approximately 0.010 in per side, leaving land fit areas at each end of the bore.
- The length of the land fit on the discharge side should be approximately 15 percent of the bore length, while the suction side land should be approximately 7.5 percent of the bore length.
- previously stated. This results in a significant differential fit, which allows a great portion of the inherent stresses to relieve during stacking.
- Record phase related runouts on the face and OD of the suction eye before removing from the boring mill. Runout of the impeller after mounting on the mandrel or the shaft should be within 0.001 in TIR of that recorded in the boring mill, accounting for all phase differences.
- Stack the impeller on the shaft vertically, and allow it to grab the shaft while resting against the spacer sleeve or shoulder. Never

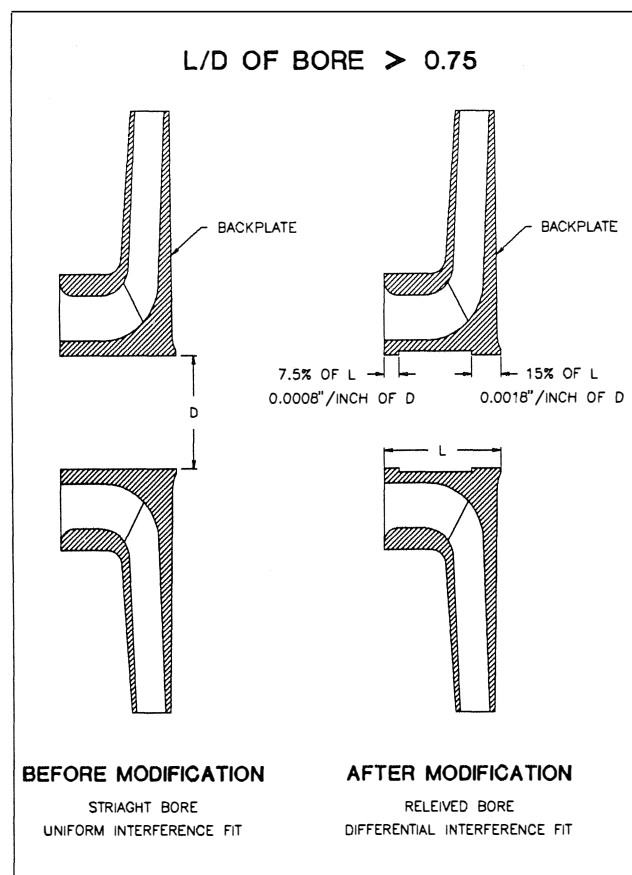


Figure 4. Modification of Impeller Bores to Reduce Stresses.

beat on an impeller with a soft hammer to “straighten” it during assembly. The bore must grab naturally, or excessive stresses will likely be induced.

- Monitor shaft deflection during stacking, and limit it to 0.0003 in TIR, accounting for any phase differences, for each component stacked.

- After the impeller has cooled and runouts are recorded, rapidly heat the suction side of the impeller to 350°F to 400°F while rolling in the balancing machine, and continue rolling until cool. This procedure helps to further minimize stresses by allowing the impeller bore to assume its normal position on the shaft, and will also allow excessive shaft deflection to straighten.

The amount of interference fit stated in the above procedure is generally applicable to those impellers with tip speeds of 750 ft/sec and above. With slower impeller tip speeds, the author has successfully modified many impeller bores in a different fashion. As previously stated, the suction side bore of a closed impeller design will open very little at speed, when compared to the discharge side. Therefore, if calculations reveal that the discharge side of the bore requires an interference fit of only 0.001 in/in diameter or less, the suction side can be made to have approximately 0.0016 in/in of diameter interference. On a six inch bore diameter, for example, the discharge side may be 0.006 in tight, while the suction side may be 0.010 in tight. This will fairly well assure that the suction side stays anchored to the shaft at all times, and allows the discharge side to slip when heavy stresses are induced. This method will also alleviate the problem of impellers “walking” down the shaft toward the suction end, a problem that commonly occurs on many flexible shaft machines that are built without differential amounts

of shrink fit in the impeller bores. If this problem occurs, the rotor loses all axial clearances between major components, which can create very high vibrations within the bending critical.

To summarize this discussion, never allow a component to deflect the shaft centerline more the 0.0003 in TIR during stacking, accounting for any phase differences that may occur.

Further, watch for excessive unbalance after a component has been stacked. Either (or both) of these are symptoms of excessive stresses that can cause real problems in the first bending critical. As a general rule, a component that has been properly mandrel balanced and assembled to the shaft will not result in an unbalance greater than approximately 8W/N.

Rough Handling of Stacked Rotors

As suggested in the previous discussion, it is impossible to stack a rotor with zero stresses. When stacked rotors with considerable bearing spans are handled roughly, such as during handling and transportation, one or more of the component shrink fits may slip and create shaft deflection. This is probably the most frequently encountered “centerline set” that occurs on stacked rotors.

As mentioned earlier, a centerline set or slight bow in the rotor, may be impossible to detect with conventional methods, such as dial indicators. This is sure to happen when rotors are transported from storage to the field, either from stresses relieving during transportation, or from simple sag due to gravity. Too many people are quick to “check balance” a rotor that has been bounced around halfway across the country.

Naturally, the “check balance” will reveal that the rotor is significantly out of balance, so some well meaning worker will “correct” it, never realizing that the rotors centerline has shifted. Then, when the rotor is installed and the centerline straightens itself during startup, everyone wonders why the vibration level is higher than it should be!

The very best policy for rotors that have been properly assembled and progressively balanced is to never “check balance” it again. There are very few turbomachinery rotors that can be transported for even short distances without developing some degree of centerline shift.

Excessive Top Clearances on Component Keys

The problem of excessive key clearances is one that is too often overlooked, and the guilty parties often include the OEM. One must remember during rotor assembly that unbalance equals eccentricity times mass. When a component key (mass) has excessive top clearance that will allow movement (eccentricity) during operation, this may be enough to cause undesirable vibration levels on a balance sensitive rotor. For example, a 7/8 in square impeller key 6.0 in long, would weigh approximately 563 grams. If the key had 0.050 in top clearance, then resultant unbalance during operation would be $0.05 \times 563 = 28.15$ gr-in, or approximately 1.0 oz-in. The author has seen key clearances of 0.100 in from the OEM!

To check key clearance, install the key in the keyway of the shaft and measure across the shaft diameter to the top of the key with an outside micrometer (Figure 5). Then, measure from the top of the keyway in the bore, across the bore to the other side with an inside micrometer (Figure 6). Add to the ID measurement the amount of interference fit, then subtract from this total the OD measurement taken off the shaft. It is very important to have some top clearance (0.004 in to 0.006 in) to prevent the installed component from hanging on the key when installed and, therefore, causing undue stresses. The one step that many fail to do is to allow for the amount of interference fit. The measurement from the opposite side of the wheel bore to the top of the keyway will increase by the amount of the interference when installed.

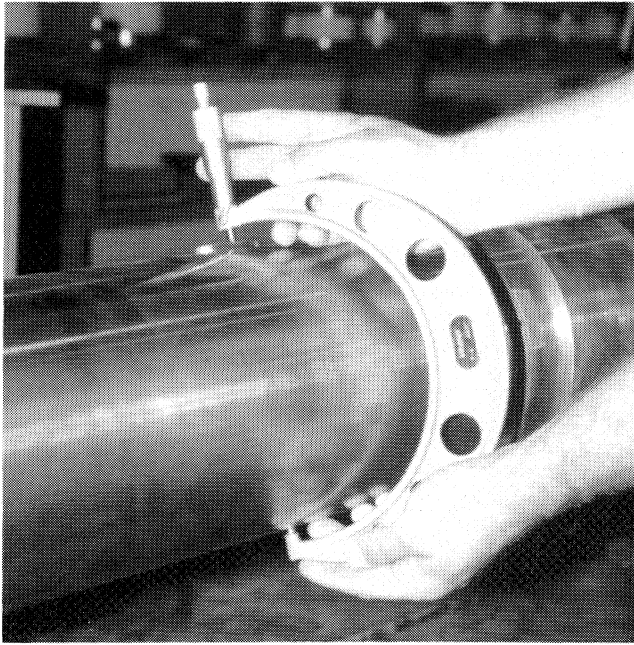


Figure 5. Measurement of Key In Shaft Dimension.



Figure 6. Measurement of Keyway and Bore Dimension.

Correction of Unbalance in Improper Planes

This problem usually occurs in conjunction with other misunderstood aspects of rotor balancing. It seems to happen every time

a properly assembled and balanced rotor is “check balanced” after developing a centerline shift from handling and transportation. It also occurs most every time a flexible shaft, high speed, multistage rotor that was operating beautifully is pulled during a turnaround for “check balancing.”

The author was once involved (after the fact!) in such a case. An 11000 lb nine stage steam turbine (Figure 7) had run for quite a few years at low vibration levels. The decision was made to pull the upper case half during a scheduled turnaround to repair leaking steam seals. But once the lid was removed, there sat the rotor just begging to be “check balanced,” so the maintenance people couldn’t stand it! They had to pull the rotor and put it in the OEM shop for “balancing.” Needless to say, the rotor shook badly during startup, at which point the author was called in to deduce that the rotor was significantly unbalanced! Efforts at inplace balancing were rewarded with limited success due to inability to accurately determine the true mode shape of the rotor and inaccessible balance correction planes. This turned out to be a very expensive balance job indeed!

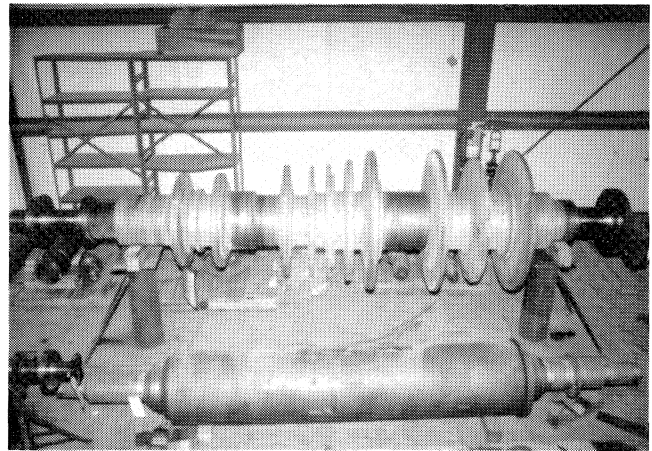


Figure 7. Flexible Rotor With Balance Problems.

Excessive Unbalance Left in Rotors During Balancing

This problem usually occurs with an “unconsciously incompetent” balance operator who assumes his machine is accurate. There are many still around that balance down to some arbitrary amount (0.1 mils, etc.) indicated by the balancing machine. This simply will not suffice with today’s turbomachinery rotors. Any good balancing machine can normally be calibrated to provide accuracy when a specific rotor is used. However, once a different type of rotor is installed, and/or the basic set up of the machine is altered from that used for calibration, the machine will no longer be totally accurate. In addition, many people do not routinely have their balancing machines serviced and calibrated, so the readout error may be even greater.

However, it really is not important that a balancing machine be absolutely accurate, for a wise balance operator will routinely perform a residual unbalance test for every rotor he balances. A residual unbalance test is performed in the following manner:

- Mark the rotor off in twelve equally spaced increments (every 30 degrees) for each correction plane.
- Select a trial weight that equals approximately twice the allowable residual unbalance.
- Place the trial weight on the first marked area, then run the rotor up to test speed and record unbalance amount and phase.

- Repeat the preceding step for the remaining eleven positions, being very careful to place the trial weight at exactly the same radius for each run.
- Plot the recorded data on polar coordinate graph paper, and calculate the residual unbalance amount (Figure 8).

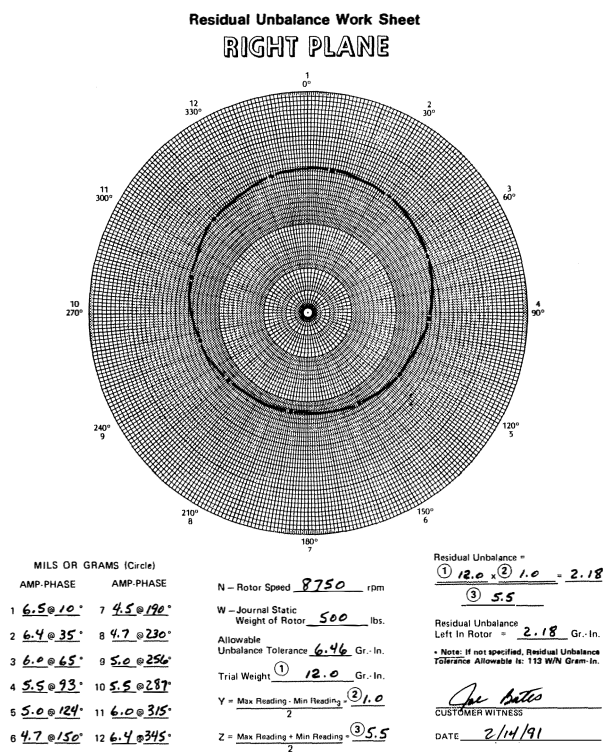


Figure 8. Example of Residual Unbalance Test.

The resultant plot should approximate a circle. If not, the most common reason is inaccurate placement of the trial weight during the test. The second most common reason for a noncircular plot is inconsistent machine readouts. A balance machine that will not read out consistently for two identical balance runs cannot be used to accurately balance or to accurately determine residual unbalance. Therefore, consistency of readouts is the most important requirement of any balancing machine.

Balancing on Areas Other Than the Bearing Journals

This is yet another common mistake that is made far too often, usually to avoid getting balancing roller marks on someone's shiny bearing journal. The intent may be honorable, but the result usually is not due to eccentricity of those areas adjacent to the bearing journals. As stated earlier, unbalance equals eccentricity times mass. It is quite common, even on new shafts, to be able to measure a distinct eccentricity on other areas of the shaft while it is supported at its bearing journals on vee blocks. Therefore, balancing should be done while the rotor is supported on its bearing journals.

CONCLUSION

Though balancing is probably the simplest aspect of turbomachinery maintenance and reliability, many problems have occurred due to inexperience, oversights, and lack of sound judgement when rotors are assembled and shop balanced. The most common mistakes have been outlined to, hopefully, increase the reader's awareness of these potential problem areas, and thereby, increase machinery reliability in general.